

Analysis of operation of the gas turbine in a poligeneration combined cycle

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Abstract In the paper the results of analysis of an integrated gasification combined cycle IGCC polygeneration system, of which the task is to produce both electricity and synthesis gas, are shown. Assuming the structure of the system and the power rating of a combined cycle, the consumption of the synthesis gas for chemical production makes it necessary to supplement the lack of synthesis gas used for electricity production with the natural gas. As a result a change of the composition of the fuel gas supplied to the gas turbine occurs. In the paper the influence of the change of gas composition on the gas turbine characteristics is shown. In the calculations of the gas turbine the own computational algorithm was used. During the study the influence of the change of composition of gaseous fuel on the characteristic quantities was examined. The calculations were realized for different cases of cooling of the gas turbine expander's blades (constant cooling air mass flow, constant cooling air index, constant temperature of blade material). Subsequently, the influence of the degree of integration of the gas turbine with the air separation unit on the main characteristics was analyzed.

Keywords: Integrated gasification combined cycle; Gas turbine; Natural gas and syngas fuel mixture; Thermodynamic analysis; Change of load

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Nomenclature

A	–	area, m^2
ABR	–	air bleed ratio
c_p	–	specific heat, $\text{J kg}^{-1}\text{K}^{-1}$
LHV	–	low heating value, J kmol^{-1}
CMV	–	compressor map variable
\dot{m}, \dot{n}	–	mass flow rate, kg s^{-1} , kmol s^{-1}
\bar{m}_z	–	reduced relative mass flow rate
N	–	power, MW
p	–	pressure, MPa
R	–	individual gas constant, $\text{J kg}^{-1}\text{K}^{-1}$
St	–	Stanton number
T, t	–	temperature, K, $^{\circ}\text{C}$
u	–	proportion of syngas in the fuel mixture
w	–	velocity, m s^{-1}

Greek symbols

α_{fg}	–	convective heat transfer coefficient for flue gases, $\text{W m}^{-2}\text{K}^{-1}$
β	–	pressure ratio
δ	–	cooling air ratio
δ^*	–	cooling air ratio for stage
ε_f	–	film cooling effectiveness, %
ε_0	–	effectiveness of blade cooling, %
η_{cl}	–	cooling efficiency, %
η_g	–	generator efficiency, %
η_i	–	isentropic efficiency, %
η_m	–	mechanical efficiency, %
κ	–	isentropic exponent
λ	–	heat transfer coefficient, $\text{W m}^{-1}\text{K}^{-1}$
μ	–	dynamic viscosity coefficient, Ns m^{-2}
ρ	–	density, kg m^{-3}

Subscripts

$1a, 2a, 1f$	–	characteristic points in the system (Figs. 1 and 2)
ASU	–	air separation unit
bl	–	blade
C	–	compressor
CC	–	combined cycle
cl	–	cooling air
cs	–	cross-section of the inter-blade channel
des	–	design condition
fg	–	flue gases
GT	–	gas turbine
in	–	inlet
out	–	outlet

- SC* – steam cycle
T – gas turbine expander

1 Introduction

Currently, the integrated gasification combined cycle (IGCC) systems are considered to be a technology that can significantly contribute to increasing the efficiency of electricity production while meeting the requirements imposed by environmental standards, including the significant reduction of carbon dioxide (CO_2) emissions [1,2]. In the future, IGCC technology may become an important competitor with other coal technologies, including supercritical coal-fired power plants, that currently dominate the area of energy technologies in the countries where coal is the dominant fuel [1,3,4].

At present, the development of IGCC units is directed toward integration with carbon capture and storage (CCS) installations. In this context, the popularity of this technology will grow primarily because of the potential for using a low-energy-consuming precombustion carbon capture technology [2]. The big advantage of such solutions is the possibility of the production in one unit of electricity and additionally many other useful products such as syngas, hydrogen, carbon dioxide, nitrogen, oxygen, compressed air, process steam, heat. The consumers of such products may be industrial plants, mainly with the chemical profile or in the case of heat – district heating systems. Polygeneration is characterized by high efficiency of transformation of fuel energy and allows the flexibility of the production cycle due to the current demand of the products in the period. An example of polygeneration IGCC can be a block, of which the construction was planned in Kedzierzyn Kozle (Poland), in which the products obtained within the IGCC system was intended to produce methanol and urea.

The main components of the IGCC unit are as follows: gasifier, synthesis gas cleaning system, gas turbine, heat recovery steam generator (HRSG) and the steam turbine. Figure 1 shows a diagram of a representative system additionally equipped with an air separation unit (ASU), which produces the oxygen needed for the gasification process, and a CCS installation, which separates carbon dioxide from the gaseous fuel (so-called precombustion carbon capture technology). Precombustion CCS requires the use of a shift reactor, in which the CO reacts with the H_2O to CO_2 and H_2 [5]. The gas turbine installation and the steam cycle that are included in the combined cycle of the IGCC unit, presented in Fig. 1, are the subjects of research presented in this paper. In the scheme the products which can be used in

the industrial unit were marked.

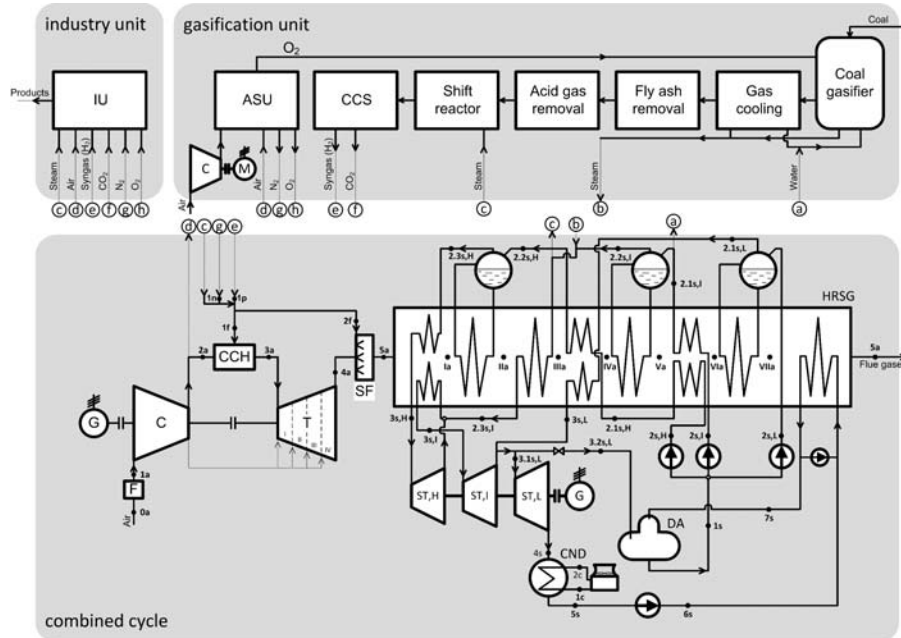


Figure 1. Scheme of the integrated gasification combined cycle (IGCC) system: ASU – air separation unit, C – compressor, CCH – combustor chamber, CCS – carbon capture and storage, CND – condenser, DA – deaerator, F – filter, G – electric generator, HRSG – heat recovery steam generator, M – motor, ST – steam turbine (H – high-, I – intermediate-, L – low-pressure part), SF – supplementary firing system, T – turbine.

In the IGCC power plant, the production and purification of synthesis gas can be realized in several ways [6–8]. The composition of synthesis gas depends on many factors, namely composition of the gasified coal, selection of the coal gasification technology, characteristics of the gasification agent and the organization and effectiveness of the gas purification processes [9]. The variety of possibilities of production and preparation of the gas ultimately fed into the gas turbine combustion chamber essentially results in the lack of possibility of equipping of the power plant with the optimal solution (with respect to the characteristics of the fuel) of the gas turbine. This difficulty arises because, despite the growing interest in IGCC technology, gas turbines are still designed mainly to burn natural gas. The transition to a fuel with different characteristics and, specifically, a fuel with a much

lower heating value and a smaller concentration of hydrocarbon compounds, results in significant changes in the operating characteristics of these machines.

In IGCC plants that use an oxygen gasification process, separation units are used for oxygen production. The most widely used technology is based on the cryogenic distillation process. In this case, the synthesis gas is produced with a very low nitrogen content, which results in a higher heating value of the flue gases. Combustion of H₂-rich gas in the gas turbine leads to an increase of flame temperature and, consequently, to an increase of temperature of blades and an increase in NO_x emission. To prevent this, and for the stability of the combustion, the synthesis gas is mixed with, e.g., nitrogen which is the product of the process of oxygen production or steam injected into the combustion chamber.

A big advantage of polygeneration units results from the possibility of changes in load and generation according to the profile of demand for these products. The gas produced in gasification unit may be used in smaller or larger amounts in nonenergy systems thus, reducing the capacity of electricity generation. Another reason for the variable availability in time of the synthesis gas may be the low availability of gasification unit. The variable demand in the year, assuming only the use of syngas for electricity generation may lead to conflicts of interests on the line energy producer – system operator. For the independence of electricity production from synthesis gas availability natural gas can be used as a fuel, which can be mixed with synthesis gas in different proportions. The composition of fuel gas fed to the gas turbine, in turn, may result from the proportions in which the synthesis gas is mixed with nitrogen or steam.

This paper presents the results of calculations based on in-house algorithm that describes the characteristics of a turbine in which synthesis gas is burned (in a gas turbine designed for natural gas) [10]. The analysis includes only a gas turbine, assuming however, that the turbine is a component of an IGCC plant equipped with CCS. In future research, the algorithm will be used in the multicriterial analysis and optimizations of IGCC plants, specifically for systems integrated with membrane CCS installations [11–15].

2 Characteristics of the gas turbine installation

The results of the analysis presented in this article refer to a gas turbine characterized by the nominal values summarized in Tab. 1. These parameters do not relate to any particular machine but may be representative of

models of gas turbine units that are currently offered by manufacturers for operation in IGCC systems. A diagram of the gas turbine model is shown in Fig. 2.

Table 1. The design-point data of the gas turbine.

Characteristic quantity	Symbol	Unit	Value
Pressure ratio	β_C	–	17
Combustor exit temperature	t_{3a}	°C	1450
Compressed air flow	m_{0a}	kg/s	525
Isentropic efficiency of the compressor	η_{iC}	–	0.89
Isentropic efficiency of the respective stages of the expander	η_{iT}	–	0.91
Mechanical efficiency of the compressor	η_{mC}	–	0.97
Mechanical efficiency of the expander	η_{mT}	–	0.97
Efficiency of the generator	η_g	–	0.985
Cooling air ratio	δ	–	0.2
Cooling air ratio of the first-stage blades	δ_I	–	0.50
Cooling air ratio of the second-stage blades	δ_{II}	–	0.35
Cooling air ratio of the third-stage blades	δ_{III}	–	0.15
Cooling air ratio of the fourth-stage blades	δ_{IV}	–	0.00
Relative inlet pressure drop	ξ_{0a-1a}	–	0.007
Relative combustor chamber pressure drop	ξ_{2a-3a}	–	0.03
Exhaust flue gas pressure	p_{4a}	kPa	105

It was assumed that a compressor is coupled with the model gas turbine, for which the characteristics of the basic parameters are shown in Fig. 3. The presented compressor characteristics were obtained from the GateCycle software [16]. The characteristics include the pressure ratio, compressor isentropic efficiency and compressor map variable (*CMV*) indicator. The *CMV* indicator defines a position of the compressor operating point in relation to the surge line and may reach values from 0.5 to 1. These parameters were obtained for different values of the reduced relative mass flow of the compressed air, which is defined as [17]

$$\bar{m}_z = \frac{\dot{m}_{1a}}{(\dot{m}_{1a})_{des}} \sqrt{\frac{T_{1a}}{(T_{1a})_{des}} \frac{(p_{1a})_{des}}{p_{1a}}} . \quad (1)$$

It was also assumed that the gas turbine is equipped with a four-stage expander. The blades of each stage are cooled in an open cycle with air taken from behind the compressor. The cooling air flows through a properly organized system of interblade channels, which allows the material of the blades

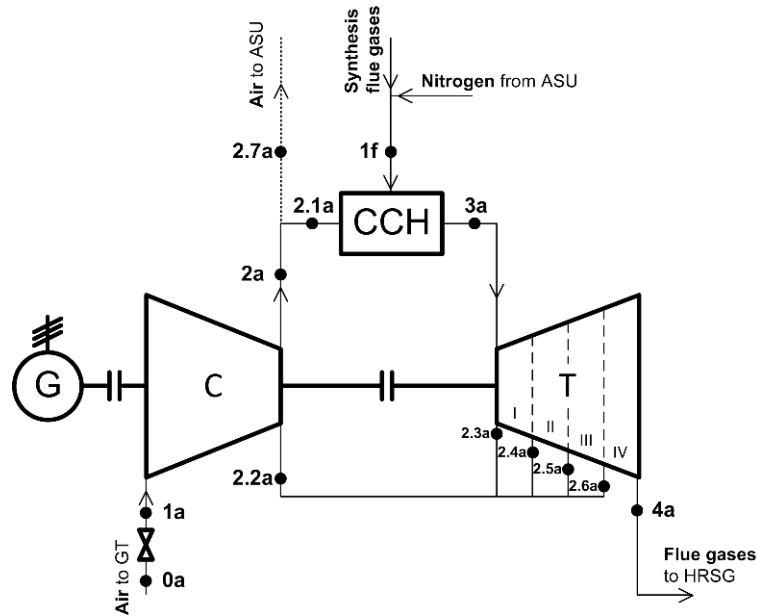


Figure 2. Diagram of a gas turbine installation.

to be maintained at safe temperature levels [18]. After cooling the blades, the air merges with flue gases flowing through the expander. In the turbine model, a simplified method of distributing the cooling air is used that reflects the fundamental characteristics of a real machine [17]. The simplification consists of a division of the stream of cooling air into a maximum of four streams, which are directed successively to the four stages of the gas turbine. The required mass flow rate of cooling air depends primarily on the flue gas temperature behind the combustion chamber, on the blade material and on the structure of the interblade channels. Significantly, a pressure ratio is also realized in the compressor, which effects the coolant temperature and thus the efficiency with which the expander blades are cooled.

In this analysis, the flow of cooling air during nominal operating conditions was determined by the cooling air ratio δ . This quantity is defined as a ratio of the stream of air directed to cooling to the mass flow rate of air supplied to the compressor

$$\delta = \frac{\dot{m}_{\text{Air to cooling}}}{\dot{m}_{\text{Air to C}}} = \frac{\dot{m}_{2.2a}}{\dot{m}_{1a}}. \quad (2)$$

The air flow directed to the cooling of the gas turbine expander's blades

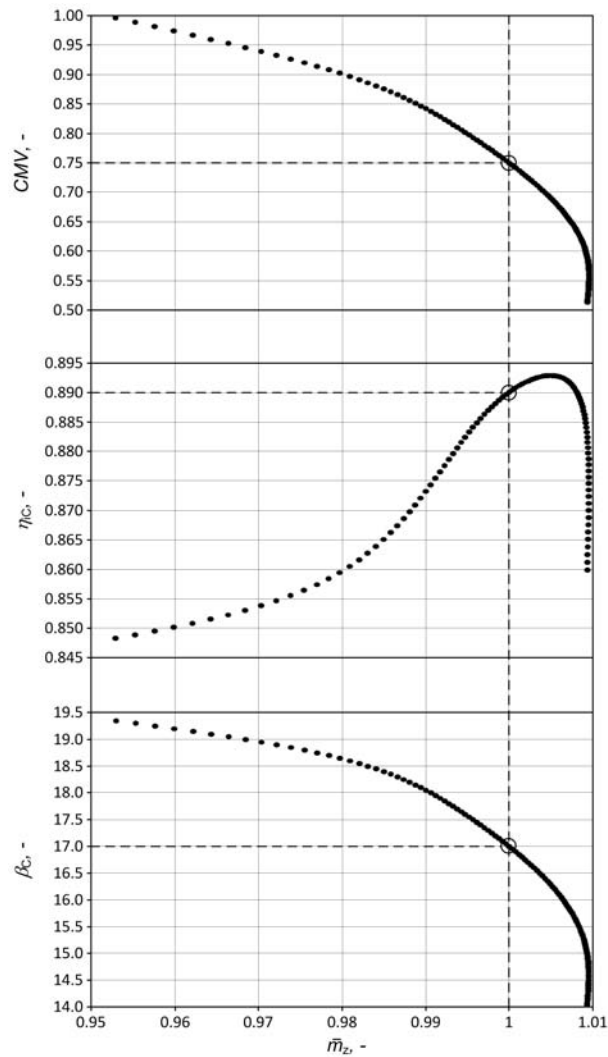


Figure 3. Characteristics of the compressor.

is divided into separate streams, which are used in various stages. In the model of the gas turbine, the mass flow rate of each stream of cooling air is determined by the assumed ratios of the mass flow rate of cooling air at each stage to the total cooling air mass flow rate

$$\delta_i^* = \frac{\dot{m}_{\text{Air to cooling of stage } i}}{\dot{m}_{\text{Air to cooling}}} = \frac{\dot{m}_{\text{Air to cooling of stage } i}}{\delta \dot{m}_{\text{Air to C}}} \quad (3)$$

Generally, the largest air streams are supplied to the blades that operate in the zone with the highest gas temperatures; thus, the blades operating within the last stages might not be cooled at all.

In IGCC units, the air compressed by the compressor, C, can also be used to supply air to the ASU installation. Because of the large amount of air required for the production of oxygen, the air is primarily compressed by independent external compressors. However, the integration of the gas turbine compressor can be justified here, and the degree of integration is described by two indicators [19–21]:

$$\text{air bleed ratio} = \frac{\dot{m}_{\text{Air to ASU}}}{\dot{m}_{\text{Air to GT}}}, \quad (4)$$

and

$$\text{integration degree} = \frac{\dot{m}_{\text{Air to ASU}}}{\dot{m}_{\text{Total air to ASU}}}. \quad (5)$$

With regard to gas turbine operation, the more important indicator is the air bleed ratio (*ABR*).

All calculations were carried out at ISO ambient parameters ($t_{0a} = 15\text{ }^{\circ}\text{C}$, $p_{0a} = 101.325\text{ kPa}$, $\varphi_{0a} = 60\%$). The nominal fuel for the modeled gas turbine is natural gas composed of (CH_4) = 0.9733, (C_2H_6) = 0.0081, (C_3H_8) = 0.0046, (C_4H_{10}) = 0.0026, (CO_2) = 0.0028 and (N_2) = 0.0086. It was assumed that the fuel is at a pressure that allows it to enter directly into the combustion chamber.

3 Model of the gas turbine

The gas turbine model allows for the simulation of machine operating characteristics away from the nominal load conditions. The nominal load (nominal operation point) is considered to exist when the turbine is powered with natural gas at ISO ambient conditions. It is assumed that at the nominal operating point, the compressor vanes are set to allow for maximum air flow. The compressed air is directed into the combustion chamber, while some is diverted to serve as a coolant for the blades of the expander. In addition, the nominal operating condition uses the values specified in Tab. 1.

In the computational algorithm, the calculations for the nominal operating condition are carried out in the first step. This step requires the determination of the mass flow rates and the thermodynamic parameters at all the characteristic points of the gas turbine. After changing the operating

conditions of the machine, the values of these parameters change. Because the gas turbine is operated with fuel other than the fuel it was designed for, the gas flow conditions change inside the turbine in response to changes in fuel flow, which changes the flue gas stream flowing through the expander. In our algorithm, the changes to the characteristic parameters of the gas turbine that accompany changes to the flow condition in the expander are realized according to the following equation:

$$\frac{\dot{m}_{fg} \sqrt{T_{fg} R_{fg}}}{p_{fg} \sqrt{\kappa_{fg} \left(\frac{2}{\kappa_{fg} + 1} \right)^{\frac{\kappa_{fg} - 1}{\kappa_{fg}}}}} = const, \quad (6)$$

where \dot{m}_{fg} is the flue gas stream flowing through a specific stage of the expander, kg/s; R_{fg} is the individual gas constant for the flue gases flowing through a specific stage of the expander, kJkg⁻¹K⁻¹; p_{fg} is the flue gas pressure at the inlet to a specific stage of the expander, Pa; κ_{fg} is the isentropic exponent for the flue gases at the inlet to a specific stage of the expander.

In the first step of the calculations, i.e., the calculations of the turbine at the nominal operating point, coolant streams brought to specific stages of the expander are determined by the assumed ratios δ_i (see Tab. 1). Thus, the analytical method for determining the amounts of these streams was not applied in the model. Nevertheless, the energy balance equations for the coolant streams should be written as follows:

$$\dot{Q} = A_{bl} \alpha_{fg} (T_{fg,in} - T_{bl}), \quad (7)$$

$$\dot{Q} = \dot{m}_{cl} c_{p,cl} (T_{cl,out} - T_{cl,in}), \quad (8)$$

where A_{bl} is the blade surface area, m²; $T_{fg,in}$ is the flue gases temperature at inlet to the stage, K; T_{bl} is the allowable temperature of a blade material (in the analysis, 1073.15 °C was assumed), °C; $T_{cl,out}$, $T_{cl,in}$ are the cooling air temperature at the outlet and at the inlet to a stage, K, respectively.

From Eqs. (6) and (7) the following can be written:

$$\dot{m}_{cl} = \frac{T_{fg} - T_{bl}}{T_{cl,out} - T_{cl,in}} \frac{A_{bl} \alpha_{fg}}{c_{p,cl}}. \quad (9)$$

By defining the effectiveness of blade cooling as

$$\varepsilon_0 = \frac{T_{fg} - T_{bl}}{T_{fg} - T_{cl,in}}, \quad (10)$$

and the cooling efficiency as

$$\eta_{cl} = \frac{T_{cl,out} - T_{cl,in}}{T_{bl} - T_{cl,in}}, \quad (11)$$

and substituting these values into Eq. (8), the following formula can be obtained:

$$\dot{m}_{cl} = \frac{A_{bl}\alpha_{fg}}{c_{pcl}} B, \quad (12)$$

where

$$B = \frac{\varepsilon_0}{\eta_{cl}(1 - \varepsilon_0)}. \quad (13)$$

The exhaust gas stream is described as

$$\dot{m}_{fg} = A_{cs}w_{fg}\rho_{fg}, \quad (14)$$

where w_{fg} is the flue gas velocity in the interblade channel, m s^{-1} , ρ_{fg} is the density of the flue gases in the interblade channel, kg m^{-3} .

Dividing Eq. (9) by (10), we arrive at

$$\frac{\dot{m}_{cl}}{\dot{m}_{fg}} = \frac{A_{bl}\alpha_{fg}c_{pfg}}{c_{pcl}A_{cs}w_{fg}\rho_{fg}c_{pfg}} \frac{\varepsilon_0}{\eta_{cl}(1 - \varepsilon_0)}. \quad (15)$$

Additionally, by introducing the Stanton number for the flue gases,

$$\text{St}_{fg} = \frac{\alpha_{fg}}{c_{pfg}\rho_{fg}w_{fg}}, \quad (16)$$

the following equation is obtained:

$$\frac{\dot{m}_{cl}}{\dot{m}_{fg}} = B \text{St}_{fg} \frac{c_{pfg}}{c_{pcl}} \frac{A_{bl}}{A_{cs}}. \quad (17)$$

Taking into account the film cooling in B , we have

$$B = \varepsilon_f + \frac{\varepsilon_0 - \varepsilon_f}{\eta_{cl}(1 - \varepsilon_0)}, \quad (18)$$

where ε_f is the film cooling effectiveness defined as

$$\varepsilon_f = \frac{T_{fg} - T_{bl,f}}{T_{fg} - T_{cl,out}}, \quad (19)$$

where $T_{bl,f}$ is the characteristic temperature for film cooling in K.

The use of Eq. (13) allows for determination of the cooling air mass flow rates required to ensure proper cooling of the expander blades. The film cooling effectiveness, ε_f , and cooling efficiency, η_{cl} , depend on the cooling system solution; thus, they generally depend on the class of gas turbine. For our analysis, values of 0.4 and 0.7 were assumed for the film cooling effectiveness and cooling efficiency, respectively.

Changes to the gas turbine operating conditions, which include changes resulting from the use a fuel other than the nominal fuel, contribute to changes in the flow conditions within the gas turbine expander. As a result of the change of fuel, increased heating of the blade material may occur, possibly damaging the blades. Therefore, to ensure safe operation of the gas turbine, it may become necessary to increase the flow of cooling air to the blades during the various operating stages [22]. Although Eq. (13) is not used in the algorithm to determine the rate of nominal coolant air flow, the equation can be modified to determine the required flow of coolant to the expander stages after the turbine load has changed from the nominal operating point. Thus, in the algorithm, the change in the coolant flow in response to loading in comparison to the nominal (design) coolant flow is determined using the following formula:

$$\frac{\dot{m}_{cl}}{(\dot{m}_{cl})_{des}} = \frac{B}{(B)_{des}} \frac{\alpha_{fg}}{(\alpha_{fg})_{des}} \frac{(c_{p_{cl}})_{des}}{c_{p_{cl}}}. \quad (20)$$

The change in coolant flow is determined separately for each stage. The relative change of the convective heat transfer coefficient present in Eq.(14) is determined using the following relationship:

$$\frac{\alpha_{fg}}{(\alpha_{fg})_{des}} = \left[\frac{\dot{m}_{fg}}{(\dot{m}_{fg})_{des}} \right]^m \left[\frac{\lambda_{fg}}{(\lambda_{fg})_{des}} \right]^{1-n} \left[\frac{(\mu_{fg})_{des}}{\mu_{fg}} \right]^{m-n} \left[\frac{c_{p_{fg}}}{(c_{p_{fg}})_{des}} \right]^n. \quad (21)$$

In modeling of the gas turbine operation, the calculations are performed in successive iterations. In the first step, the calculations are realized for the operation of the turbine supplied with natural gas. This step is the so-called design mode. The thermodynamic parameters of the air after it has been compressed are determined. Then, on the basis of the cooling air ratio (Eq. (1)), the air flow supplied to the combustion chamber is determined. On the basis of the energy balance for the combustion chamber using the assumed temperature of the exhaust gases, the natural gas mass flow rate is determined. The exhaust gases are directed to the first stage of expander. At the inlet to each expander stage, the exhaust gases are mixed with cooling air; the amount of cooling air is determined by Eq. (2). The exhaust

gases in each stage expand in response to the local pressure. In the initial phase, the pressure profile is determined by assuming a constant pressure drop in each stage. In subsequent iterations, the pressure drops are established so as to achieve a constant enthalpy drop in each stage. After using the balance equations to calculate the mass flow rates and enthalpies at each characteristic point of the gas turbine, the compressor internal power and the turbine internal power can be determined. Finally, by taking into account the mechanical efficiency and generator efficiency (Tab. 1), the power generated by the whole gas turbine is determined.

After this design mode calculation step, the parameters for the syngas-powered gas turbine are calculated. In the first step, the combustion chamber parameters are calculated. In the first iterative loop of the syngas-powered model, the air entering the combustion chamber is characterized by the parameters from the design mode calculations. For these conditions, the mass flow rate of the synthesis gas is calculated to achieve the assumed temperature of the exhaust gases. At the same time, composition of the exhaust gases is determined. The exhaust gases are mixed with cooling air before they enter subsequent stages of the expander. In the first iterative loop, the cooling air mass flow rates are determined by the assumptions that were adopted for natural gas combustion. As a result of the change in mass flow rate and composition of the exhaust gases that enter the subsequent stages, the pressure profile changes, and the new pressure profile in the expander is determined with the use of Eq. (5). As a result of the change in the pressure of the exhaust gases at the expander inlet, the compressor operating characteristics change. The new operation point for the compressor is determined by the compressor characteristics (Fig. 3). The change in the pressure ratio changes the compressed air flow. In the next computational iterative loop, the cooling air flows are changed in the expander. For this purpose, Eq. (14) is used. The B coefficient for the actual operation conditions and for the design (des) conditions are determined by Eqs. (18) and (19), the latter of which enables determination of the cooling efficiency, ϵ_0 . To determine the change in the convective heat transfer coefficient for the flue gases relative to the design condition, Eq. (14) is used. The specific heat capacities present in Eqs. (14) and (15) are determined for known gas compositions as a function of temperature. After determining the new cooling air flows and after assuming the degree of integration of the gas turbine with the ASU (Eq. (3)), the parameters for the combustion chamber are recalculated. These calculation loops are repeated until the balance equations

for each component are satisfied. In the next step, the balance equations are used to calculate the mass flow rate distributions and enthalpies at the characteristic points of the gas turbine. With this information, the compressor internal power and the turbine internal power are determined. After taking into account the mechanical efficiency and generator efficiency (Tab. 1), the power of the whole machine is then determined.

4 Results of the calculations and discussion

The following analysis primarily concerns the influence of the composition of the gaseous fuel burned in the combustion chamber of a gas turbine with its basic operating characteristics. Toward our ultimate goal of studying IGCC systems integrated with membrane CCS installation, the analysis focuses on the operation of a gas turbine powered by the synthesis gas that was produced within the following process: oxygen gasification of coal, cleaning of the primary gas, carbon dioxide sequestration and mixing of gas with nitrogen obtained as a byproduct of oxygen production in the ASU. Without going into a detailed analysis of the process involved in the production of this fuel, it was assumed in this study that the gas supplied to the combustion chamber is a mixture of two components: $(N_2) = 0.5$ and $(H_2) = 0.5$. For the analysis of the gas turbine operation, it was assumed that the turbine inlet temperature (TIT) is maintained at a constant level, i.e., 1450 °C (see Tab. 1), despite any changes in the nature of the operating conditions.

The results of analysis of gas turbine operation and whole combined cycle are shown. The gas turbine is supplied with a gas of which the composition is a result of mixing of synthesis gas with natural gas. Respectively, Sections 4.1 and 4.2 is related to a gas turbine operation, and Section 4.3 to the operation of the entire combined cycle. Despite the focus on the influence of fuel composition, the present section will focus on the degree of integration between the gas turbine and the ASU. Finally, the last paragraph of this section presents an analysis of the compressed air regulation achieved by changing the angle of the guide vanes as a method for maintaining the desired operating parameters of the gas turbine in the context of cooperation of the gas turbine with the steam part.

4.1 Effect of fuel composition

In the first step, the effect of fuel composition on the basic operating characteristics of the gas turbine was analyzed. It was assumed that the gas turbine operates in the IGCC system, where oxygen gasification of coal takes place. The raw synthesis gas is first subjected to cleaning, followed by the oxidation of CO in the gas to CO₂ and ultimately by CO₂ capture. The gas that results from this process is characterized by a very high hydrogen content. This gas, prior to entering the combustion chamber, is mixed with nitrogen, which is also a byproduct of the oxygen separation process conducted in the ASU installation. In this case, the gas supplied to the gas turbine combustion chamber is characterized by high hydrogen and nitrogen content. The main aim of this study was to determine the transition characteristics of the operating parameters resulting from the change of fuel from natural gas to synthesis gas. Fuel composition was changed according to the u index, defined as follows:

$$u = \frac{\dot{n}_{\text{synthesis gas}}}{\dot{n}_{\text{natural gas}} + \dot{n}_{\text{synthesis gas}}}, \quad (22)$$

where $\dot{n}_{\text{synthesis gas}}$ is the molar flow rate of synthesis gas ((N₂)=0.5, (H₂)=0.5), kmols⁻¹, $\dot{n}_{\text{natural gas}}$ is the molar flow rate of natural gas, kmols⁻¹. The value of the u index was changed from 0 to 1 for our analysis.

The u parameter determines the composition of gas, which is supplied to a combustion chamber of the gas turbine (Fig. 4). In the case of availability of any amount of gases for any u parameter the flow of fuel combusted in the gas turbine is determined based on modelling of the operation of the gas turbine. The change of the fuel flow is a result of a change of load of a machine leading to changes in the position of operating points of its respective components. The hypothetical situation can be also assumed in which the production capacity of the unit responsible for the generation of synthesis gas corresponds to the capacity of electricity production of the unit. In such a case the operation of the system for $u = 1$ is possible. All additional demand for synthesis gas, for example for the industry unit, results in limitation of the possibility of electricity production. In this case, for the possible supplement of production capacity in this part of the system the use of natural gas can be considered. In the extreme case, when the demand for synthesis gas for the nonenergy purposes reaches the maximum dispositional level or in the case of service of a gasification unit the combined cycle can only be supplied by natural gas. The possibility of

operation of a gas turbine supplied by a gas characterized by composition resulting from a value of the u parameter in the range from 0 to 1 may be limited by structural considerations and by obtaining the limit values for such quantities as temperature of blade material, flame temperature, the CMV parameter, or the temperature of air leaving the compressor [19].

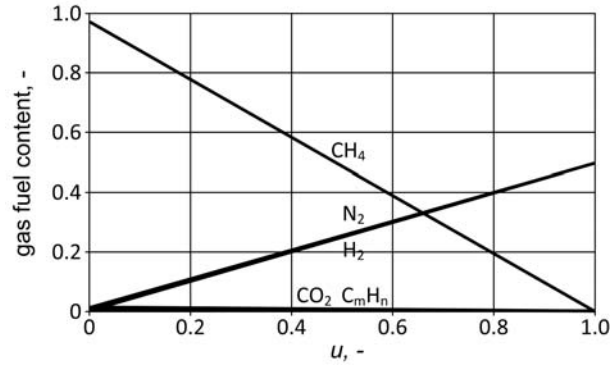


Figure 4. Composition of fuel combusted in gas turbine as a function of the proportion of synthesis gas in the fuel mixture.

In the next Section (4.2) the gas turbine is analyzed in the context of integration with an air separation unit. The degree of separation is determined by the ABR parameter defined there. For the purpose of the present point the calculations were carried out for the ABR parameter equal 0. Figure 5 presents the main characteristic quantities of operation of a gas turbine as a function the proportion of syngas u in the fuel mixture. In the plot, among others, the characteristics of the efficiency of the gas turbine are shown. This quantity is defined by the following relationship:

$$\eta_{GT} = \frac{(N_{iT}\eta_{mT} - \frac{N_{iC}}{\eta_{mC}})\eta_g}{n_{\text{natural gas}}LHV_{\text{natural gas}} + n_{\text{synthesis gas}}LHV_{\text{synthesis gas}}} . \quad (23)$$

Similarly, the effect of the proportion of syngas on the gas turbine power, compression and expansion power, compressed air stream, flue gas temperature and the index of air cooling blades of the expander are presented in Fig. 5. The figures present analyses based upon three different assumptions. The solid line represents the change in the flow rate of the cooling air that was delivered to the expander blades based on the methodology presented in Section 3 (Eqs. (8) and (10)). The dashed line represents the findings from the analysis of the case in which the cooling air index remains

constant ($\delta = const$). The dotted line represents the findings from the analysis of the case in which the cooling air mass flow rate was held constant ($\dot{m}_{1a} = const$).

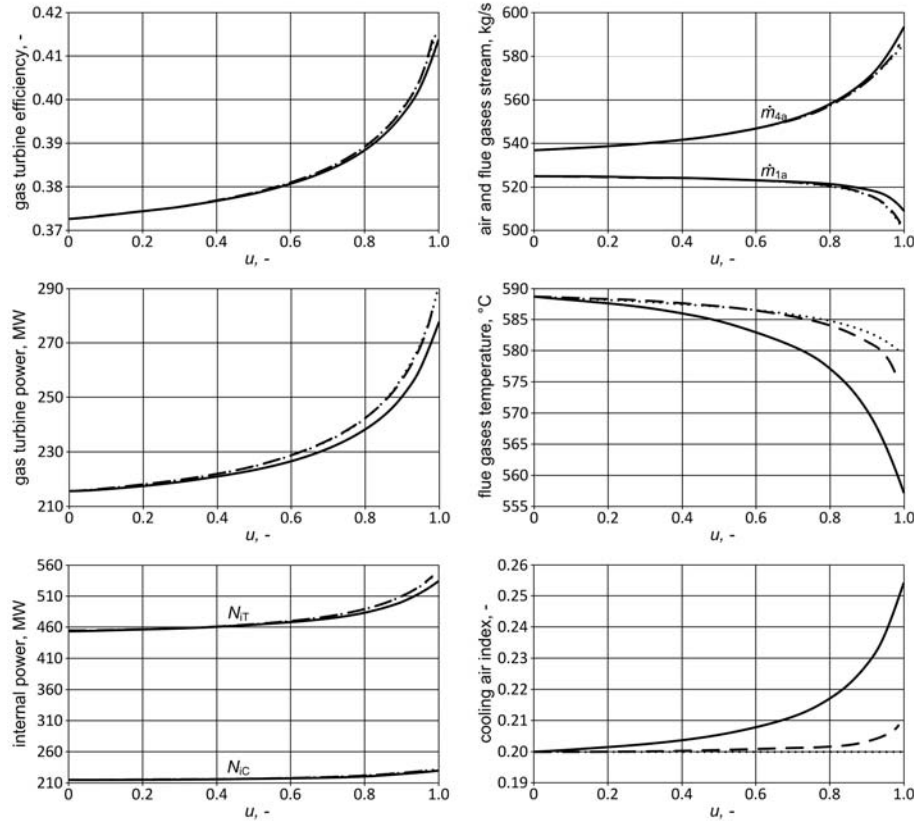


Figure 5. Characteristic quantities of gas turbine as a function of the proportion of synthesis gas in the fuel mixture.

As shown in Fig. 5, when the δ index and the cooling air flow rate \dot{m}_{1a} are held constant at the nominal level (dashed and dotted lines), the operation of the gas turbine is not possible in the high ranges of syngas input because the compressor is operating at a CMV index value equal to one, i.e., on the surge line. The transition from the designed fuel to the synthesis fuel is associated with increased efficiency of the gas turbine. Of the three cases examined, the smallest increases were recorded in the case in which the cooling air streams entering the expander were varied in accor-

dance with the methodology presented in Section 3. When the gas turbine is operating under nominal conditions, its efficiency is 0.3728; when it is powered by synthesis fuel, the efficiency is 0.4138.

Similar conclusions apply to the gas turbine power: the turbine power increases from 215.8 MW to 278.1 MW when switching the fuel to syngas. The main reason for the increased efficiency of the gas turbine is the increase of the exhaust gas stream flowing through the expander that occurs with the increasing u index. This increased exhaust flow rate results in a significant increase in the expander's power. Additionally, the decrease in the air stream compressed in the compressor contributes to the increased efficiency. This increased efficiency occurs because the decreased flow rate of the compressed air into the compressor balances the increased pressure ratio and leads to a relatively small increase in the power required to drive the compressor. The temperature of the exhaust gas leaving the expander decreases with increasing u . The use of the synthesis gas in the gas turbine leads to a decrease in the exhaust gas temperature from 588.7 to 557.1 °C. The use of synthesis gas may therefore result in a reduction in the temperature of the steam produced by a heat recovery steam generator. When a temperature level that is safe for the material of the blades is maintained, the change in the flow conditions within the expander contributes to a significant increase in the coolant streams introduced into each stage of the expander. The value of the δ ratio increased from the nominal value of 0.2 to a level of 0.2547.

4.2 The degree of integration of the gas turbine with an ASU

The next stage of our analysis was focused on the influence of the degree of integration of the gas turbine installation with the oxygen separation installation. To determine the degree of integration, an indicator defined by Eq. (3) was used. According to [19], the optimal level of integration is in the range of 0.05–0.07 to achieve a compromise between generated power and efficiency and the need to maintain the reliability. This analysis concerned only indicators of the gas turbine operation. Figure 6 shows the results of calculations for these quantities, which incorporate the analysis of the influence of synthesis gas in the fuel mixture (see Fig. 5). The degree of integration varied within the range of 0–0.1. For this analysis the u parameter was equal to 1, what means that in the combustion chamber of the gas turbine only the H_2/N_2 mixture gas was combusted.

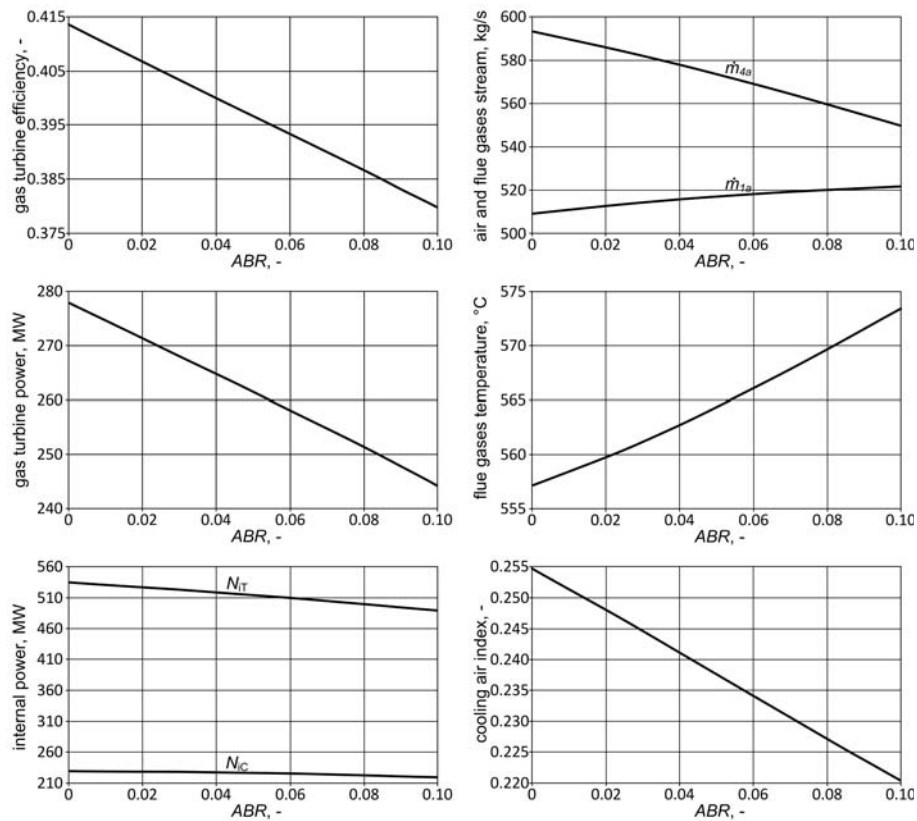


Figure 6. Characteristic quantities of gas turbine as a function of the degree of integration of a gas turbine with an ASU installation

The increase of the ABR parameter contributes to a significant reduction in the gas turbine efficiency. For an ABR parameter of 0.1, the efficiency was 0.3798, which was 0.0340 lower than the efficiency of the turbine without integration. Integration also reduces the power of the turbine, because an increase of the ABR leads to a decrease of all flows except of the compressed air. Importantly, an increase in the degree of integration translates into the increase in the flue gas temperature, which is significant in the context of producing high-temperature steam from the heat recovery steam generator. Generally, an increase in the degree of integration serves to preserve the nominal operating parameters that would otherwise occur from powering the gas turbine with a fuel other than that for which it was designed.

The performance characteristics of gas turbines using gases with com-

positions different from the nominal gas are important in the context of the role of the turbine in steam generation in the IGCC system. The temperature of the exhaust gas leaving the gas turbine expander is of particular importance because it determines the temperature of steam produced within the heat recovery steam generator.

Figure 7 shows the characteristics of two quantities, namely the temperature of flue gases at the outlet of the gas turbine and the gas turbine efficiency. These figures are shown as a function of the *IGV* parameter

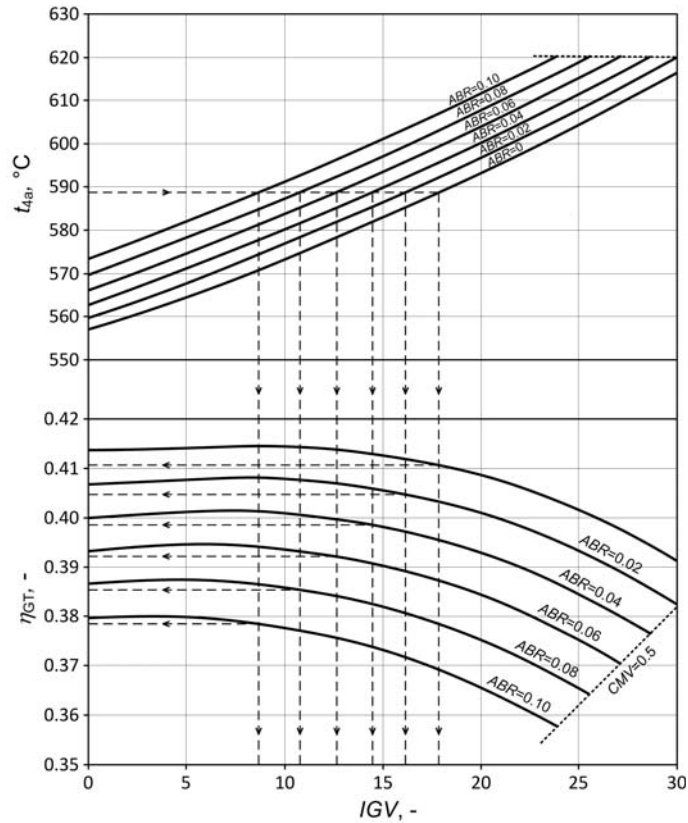


Figure 7. The gas turbine efficiency and the temperature of the flue gases as a function of the *IGV* parameter for different *ABR* parameters.

for five values of the *ABR* parameter, which defines the degree of integration of the gas turbine with the ASU installation. In the upper part of the figure, the broken line indicates a temperature of 588.7 °C, which is the nominal exhaust gas temperature for the gas turbine operating with

natural gas. As shown in Fig. 7, this temperature level can be obtained by changing the angle of the compressor vanes. In addition, as the degree of integration of the gas turbine with the ASU installation increases, less closure of the inlet guide vanes is required to obtain the nominal exhaust gas temperature. Similarly, the variation of the gas turbine efficiency with *IGV* is presented at the bottom of Fig. 7. The inlet guide vanes (*IGV*) parameter was introduced, which determines the relative change in the air stream flowing through the compressor caused by the changes in the slope of the inlet guide vane blades. It was assumed that the *IGV* would be in the range from 0% to 30%, which indicates that the maximum closure of the guide vane reduces air flow by 30%. The calculations were realized for *u* parameter equal to 1.

5 Summary

The competitiveness of the IGCC systems results primarily from the two factors, i.e., integration possibility of the system with a relatively low-energy-demand precombustion carbon capture technology, as well as from a multiproduction of useful products such as electricity, heat, carbon dioxide, nitrogen, oxygen or synthesis gas. The recipient of many products can be chemical industrial unit. The indispensable condition in this case is the flexibility of production, thus, the work of an IGCC system according to the demand profile. During the deficit of synthesis gas, the gas turbine may be supplied with the gas being a mixture of synthesis gas and natural gas. Mixing or the use of only one of these gases can be realized in the polygeneration systems, where the synthesis gas availability is variable in time due to its use in industrial units integrated with IGCC system or service necessity of a gasification unit. The different composition of this mixture gas influences both, the nature of the conversion process within the gas turbine and the heat exchange in the heat recovery steam generator.

This paper presents the calculations made for a gas turbine installation and a steam cycle, and the results of modeling of a gas turbine of known nominal operating characteristics and powered with fuel other than the fuel it was designed for. The analysis was conducted for a gas turbine operating on a mixture of natural gas and synthesis gas, which can be produced from coal gasification in oxygen, among other sources, followed by a carbon dioxide capture process. As the results show, running the gas turbine on syngas strongly affects the operating characteristics of the gas

turbine. One way of mitigating the deviation from the nominal operating conditions is to integrate the gas turbine with the ASU installation.

Despite the higher gas turbine efficiency observed with the use of synthesis gas, the negative aspects of running the turbine in this manner should be emphasized. With regard to thermodynamics, the primary concern is the significant decrease in exhaust gas temperature at the outlet of the expander that occurs in the absence of regulation. This decrease in temperature contributes to an undesirable temperature drop in the steam produced by the heat recovery steam generator. As a part of the steam cycle of the IGCC system, the decreased steam temperature leads to a drop in the steam cycle efficiency. Changing the angle of the compressor inlet blades can prevent the exhaust gas temperature drop.

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